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(54) Thermo-acoustic refrigeration apparatus

(57) In thermo-acoustic and pulse coolers which operate through the compression and expansion of a working fluid (5) discontinuities in the laminar boundary layer flow of the working fluid over a regenerator plate 2 are caused by pads (3) on the surface of the plates. Alternatively axially spaced ribs are provided, or instead of a series of plates, the regenerator is formed from a coiled strip or nested coaxial tubular plates. The refrigerator comprises a compressor (11), a reservoir (12), and an elongate tube (1). A series of fins (14) remove heat from the tube to ambient. The evaporator portion (7) of the tube contains finely divided metallic material (9), eg wire wool. The downstream part of the tube includes regenerators (15).

Fig.1

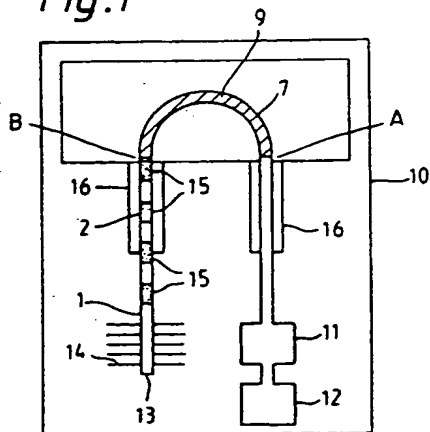


Fig.2

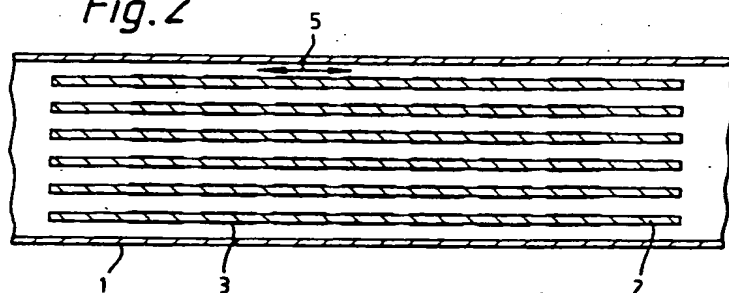


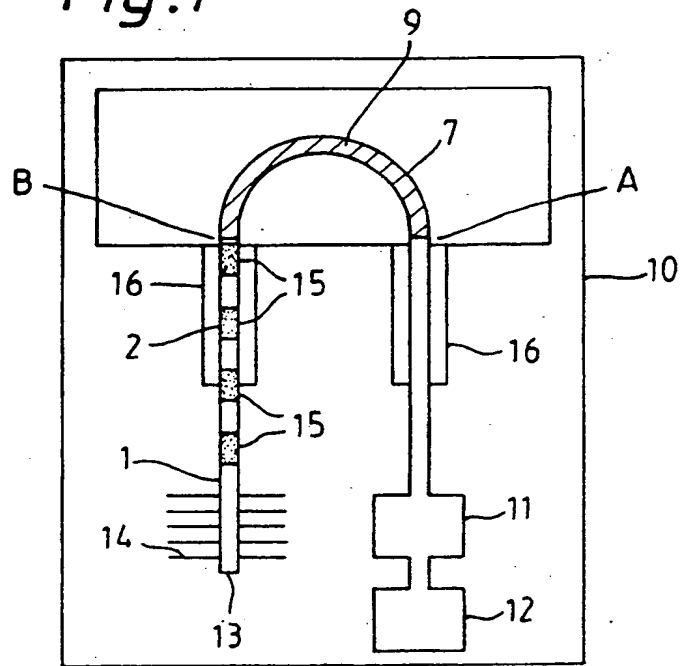
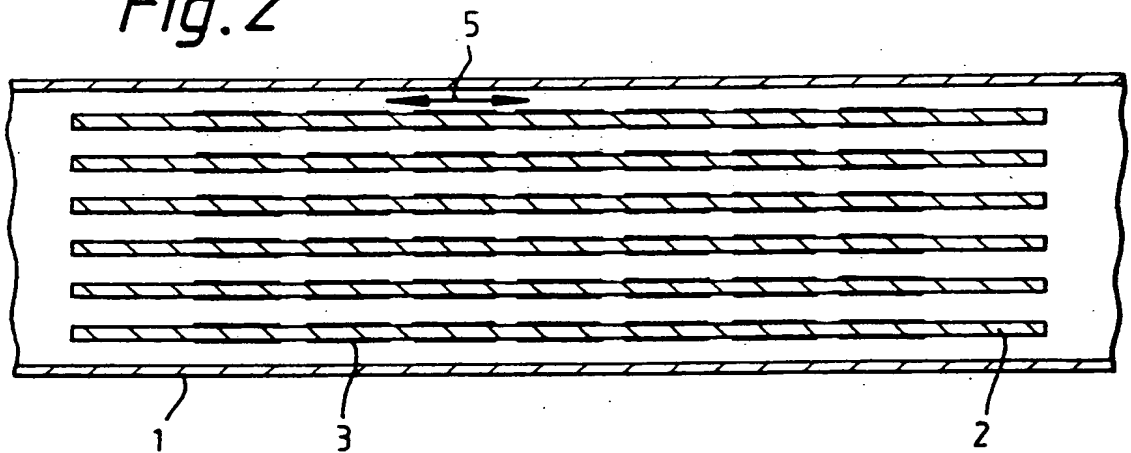
Fig.1*Fig.2*

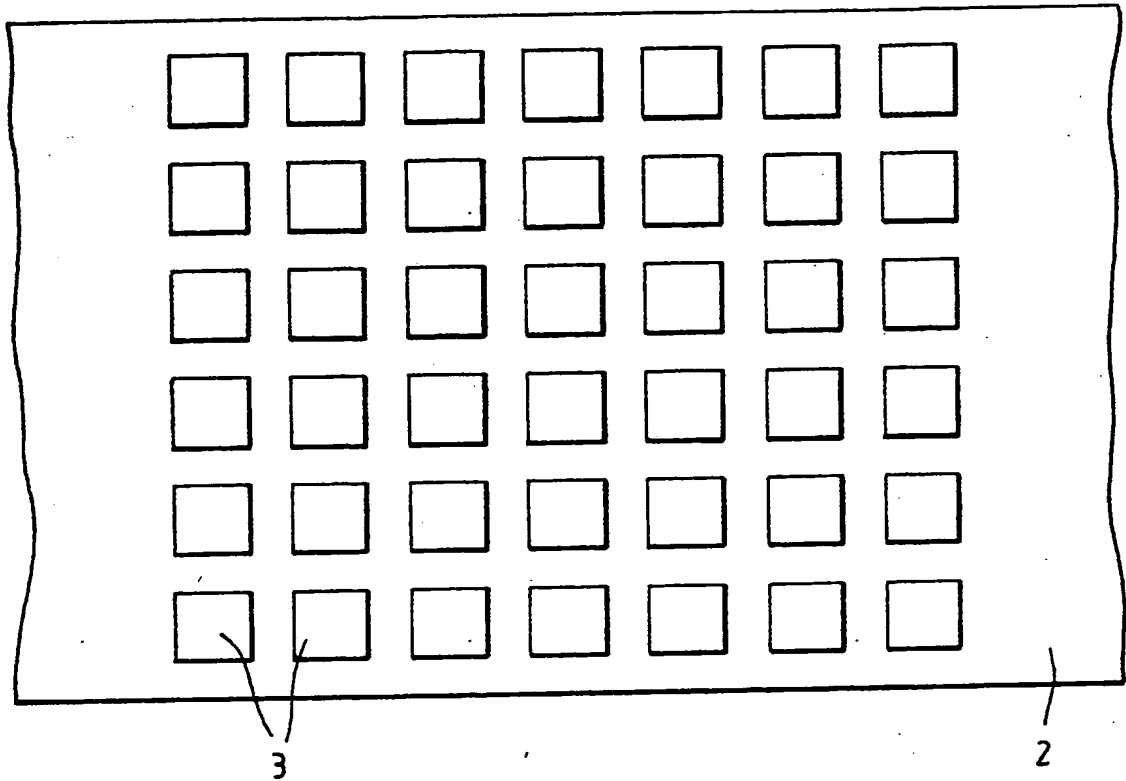
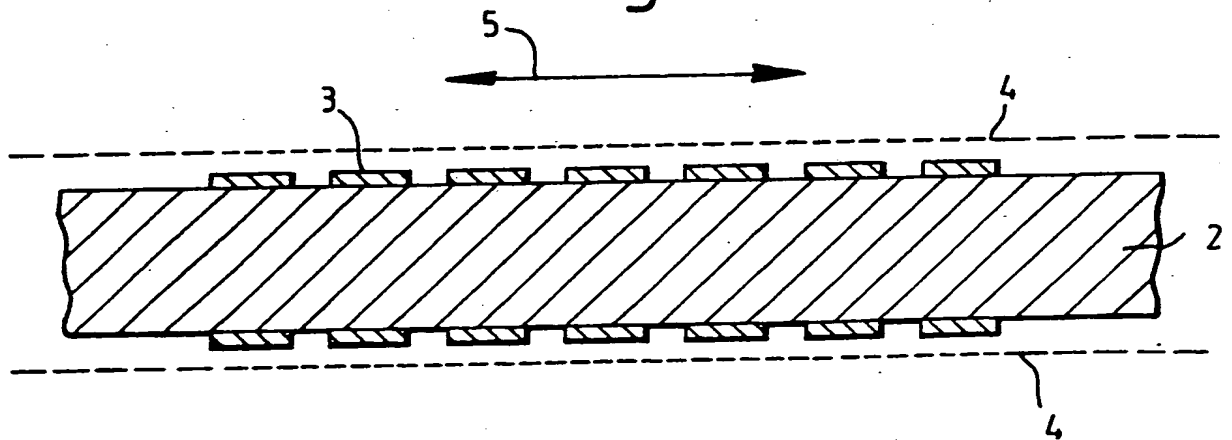
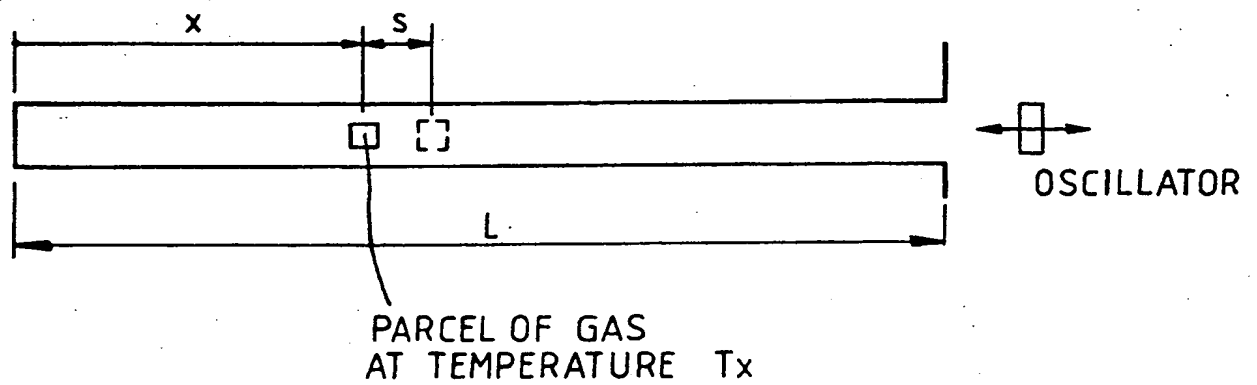
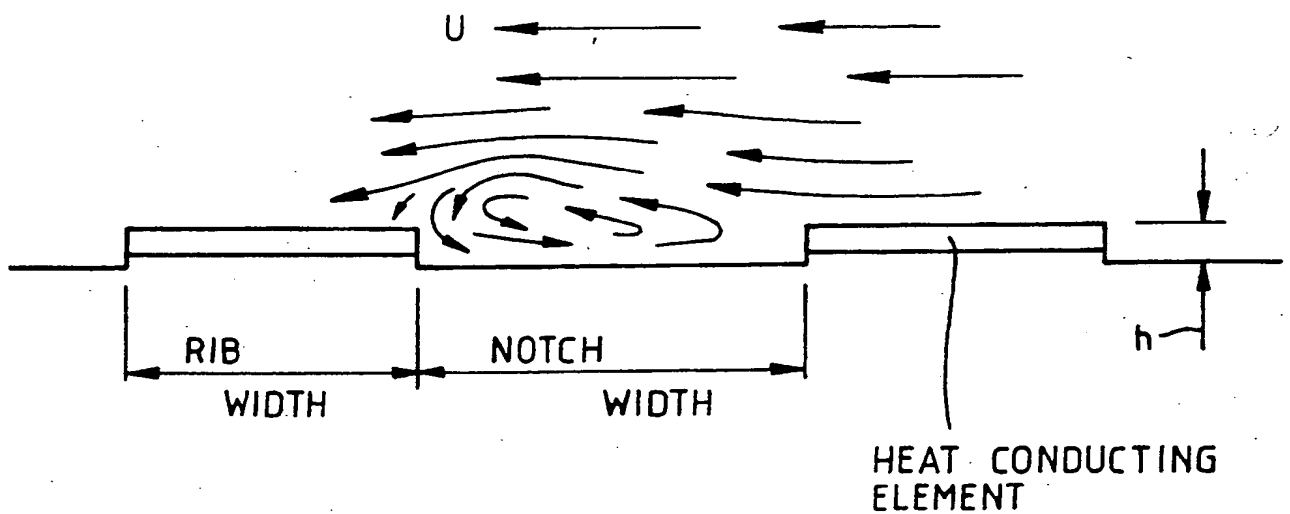
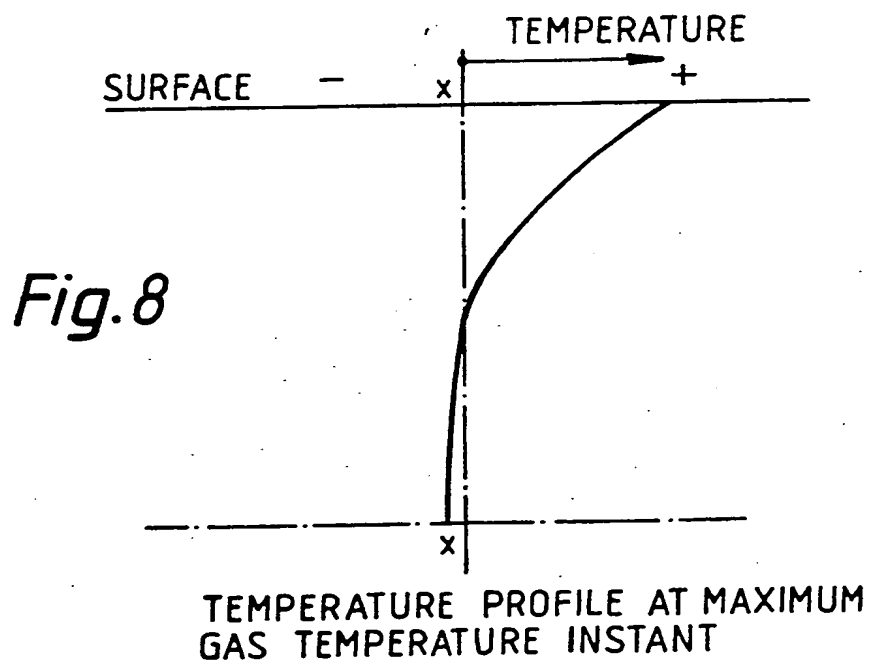
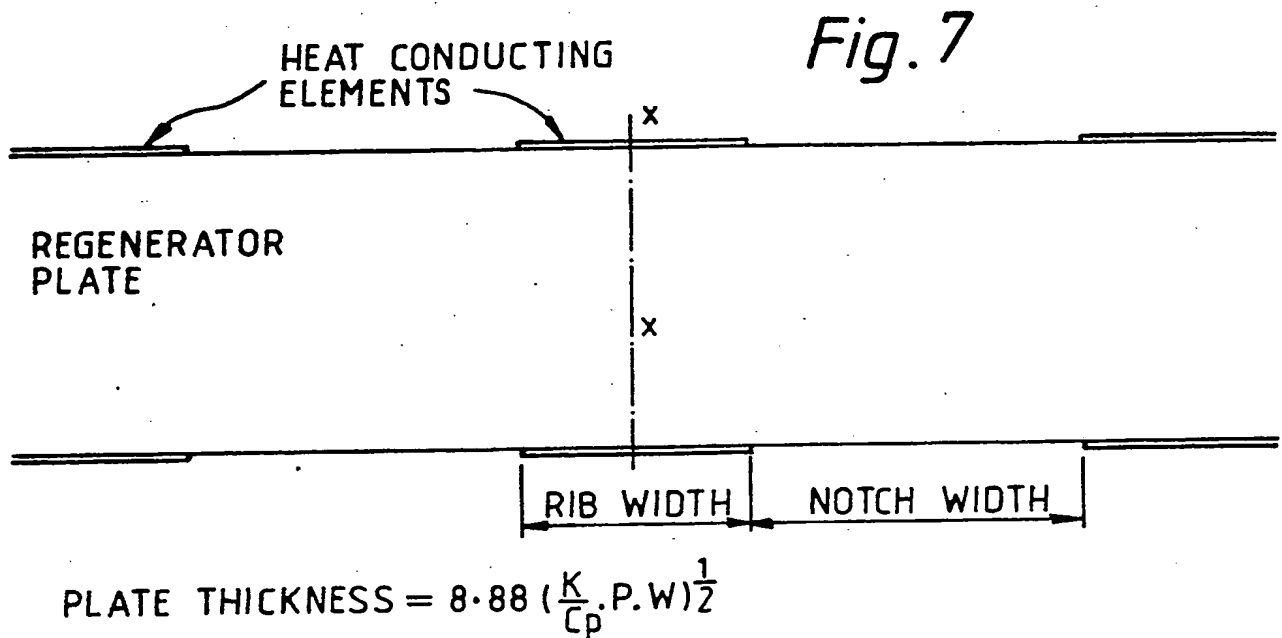
Fig. 3*Fig. 4*

Fig. 5*Fig. 6*



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REFRIGERATION APPARATUS

This invention relates to refrigeration (cooling) apparatus and is more particularly, but not exclusively, concerned with thermo-acoustic refrigeration apparatus of which the operation, and examples of the construction, are described in prior art such as:-

- 1) "Natural Engines" by J. Wheatley and A. Cox in "Physics Today", Volume 38, No 8 (August 1985);
- 2) American Society of Mechanical Engineering (ASME) paper No 63-WA-290 presented at the Winter annual meeting of the ASME in Philadelphia, Pennsylvania, USA during November 1963 by W. E. Gifford and R. C. Longworth;
- 3) "Thermo-acoustic Effects in a Resonance Tube" by P. Merkli and H. Thoman in the Journal of Fluid Mechanics, 70:161 (1975);
- 4) "Understanding Some Simple Phenomena in Thermo-acoustics with Applications to Acoustical Heat Engines" by J. Wheatley, T. Holler, G. W. Swift and A. Migliori in the American Journal of Physics, Volume 53, No 2 (February 1985); and
- 5) "A Comparison of Three Types of Pulse Tube Refrigerator: New Methods for Reading 60°K" by R. Radelbough, J. Zimmerman, D. R. Smith and B. Louuse, submitted to Advances in Cryogenic Engineering: Volume 31.

The present invention concerns both the non-resonant "pulse tube" type of apparatus forming the subject of items 2 and 5 in the above list of references and also the resonant "organ pipe" apparatus to which items 3 and 4 refer.

The resonant "organ pipe" refrigeration apparatus operates by the movement, compression and expansion of a working fluid, ie; a gas or vapour, in an enclosure in a manner somewhat similar to the behaviour of air in an organ pipe - the behaviour of the working fluid obeys the same laws of physics which pertain to air in an organ pipe apart from slight modification due to the presence of one or more structures known as regenerators which are sometimes placed inside the enclosure. For non-resonant conditions, ie; with the pulse tube refrigeration apparatus, the same laws of physics apply although the simple organ pipe analogy is not then appropriate.

The regenerator is a kind of heat exchanger through or past which the working fluid flows in a reciprocating mode and which has the function of receiving heat energy from the working fluid, temporarily storing it, and giving it up again in such a way that given the movement, compression and expansion of the working fluid, heat is transferred from one part, called the 'cold end' to a part at the other side of the regenerator. Regenerators are also used, for a similar purpos , in cooling devices operating on the Stirling Cycle.

The performance of thermo-acoustic refrigeration apparatus relies on the existence of a time phase lag between the working fluid pressure cycle and the efficient temporary storage of thermal energy by the regenerators, and its later release back to the gas. At present a regenerator would normally comprise a porous mass of metallic material, for example a stack of gauze discs or a plug of wire wool. It is believed that the use of such known forms of regenerator constitutes a limit on the achievable performance of thermo-acoustic coolers and the object of the invention is to provide a regenerator which may be improved in this respect.

According to the present invention there is provided refrigeration apparatus of the kind comprising a regenerator and a working fluid which is in contact with the regenerator and which is also subjected to movement compression and expansion so as to produce a cooling effect, said regenerator defining at least one surface alongside which said working fluid passes and which has a plurality of portions projecting out into the working fluid so as to create local controlled discontinuities in the boundary layer flow of the working fluid and hence improve heat transfer between the regenerator and the working fluid and further so as to create localised heat source/heat sink sites for increasing the thermal phase lag between said surface and the bulk of the member defining that surface,

and increasing the effective distance through which thermal energy is pumped per cycle.

For better understanding of the invention, reference will now be made by way of example, to the accompanying drawings, in which:-

Figure 1 is a diagrammatic, rear-view of a thermo-acoustic refrigerator;

Figure 2 is a sectional view of regenerator used in the figure 1 refrigerator;

Figures 3 & 4 are respectively a sectional side view and a plan view of one plate of the figure 2 regenerator;

Figure 5 depicts a pipe closed at the left hand end and in which acoustic behaviour is stimulated by an oscillator at the right hand end;

Figure 6 depicts for a transverse rib case, gas flow lines, and a primary vortex state present in the vicinity of each step-down followed by step-up pair, and associated primarily with the step-up condition; and

Figures 7 and 8 define the geometry and, as an example, depict an instantaneous temperature state within a regenerator plate.

The refrigerator of figure 1 comprises a cabinet 10 at the rear of which there is mounted a compressor 11, a fluid reservoir 12 coupled to the compressor 11, and an elongate

tube 1 of which one end is also coupled to the compressor. From the compressor 11, the tube 1 extends up and, at the position marked "A", enters the cabinet 10. The tube re-emerges from the cabinet at the position marked "B" and descends down to its other end alongside the lower rear part of the cabinet. This other end of the tube 1 is closed and, near it, the tube is connected to a series of vanes or fins 14, forming a heat exchanger for removing heat from the tube to the ambient atmosphere.

The portion 7 of the tube, between positions "A" and "B", which lies within the cabinet 10 is shown as if it is simply looped over from the position "A" to position "B" but, in reality, it would follow some appropriate path within the cabinet for receiving heat therefrom in an efficient manner. For example, the tube could follow a winding or convoluted path alongside an internal surface of the cabinet or of a special cold compartment of the cabinet and/or it could be arranged to form part of the structure of one or more shelves within the cabinet. Its cross-sectional shape need not be uniform along its length.

External portions of the tube adjacent to the cabinet entry and exit positions A and B may be insulated, for example by means of expanded polystyrene jackets 16 as shown.

The internal portion 7 of the tube 1 contains finely divided metallic material 9, for example wire wool, to assist in the collection of heat.

Meanwhile, the portion of the tube between position "B" and the heat extraction fins 14 contains four axially spaced regenerators 15 each in the form of a set of face-to-face plates 2 which will be described further with reference to figures 2,3 and 4, and in addition the entire tube 1, the compressor 11 and the reservoir 12 are filled with a suitable working fluid of a kind suitable for use in a thermo-acoustic cooler, for example Nitrogen.

The function of the reservoir 12 is to maintain the pressure of the gas within the compressor 11 and tube 1 at some appropriate mean level while the compressor 11, which may be of the type comprising a reciprocating piston (not shown) has the job of generating a cyclic pressure variation within fluid in the tube so that a standing acoustic wave is set up in the fluid along the length of the tube. As discussed earlier herein and in the aforementioned prior art references, this results in movement of the working fluid back and forth past the heat regenerators and at the same time local fluid pressure changes which together result in a net transfer of heat from the compressor side of the regenerators, ie; from the portion 7 of the tube which is inside the cabinet 10, to the other side of the

regenerators, ie; to the portion of tube 1 which is connected to the heat extraction fins 14.

As mentioned, the regenerators 15 each comprise a set of plates 2, the sets being spaced apart in the axial direction of the tube 4. Figure 2 shows one of the sets of plates in its section of the tube 1, each of the other three sets being similar to the one shown. The plates 2 in the illustrated set extend parallel to one another and to the axis of the tube 1 and are spaced apart so that the working fluid can pass between them. As shown in figures 3 and 4, each plate has on each of its faces a two-dimensional array of pads 3 which project from the face of the plate a short distance out into the laminar boundary layer 4 of the working fluid.

Instead of a series of plates, each regenerator could comprise a coiled strip of material with the turns of the coil spaced one from another for fluid to pass there between.

Also instead of the heat conducting pads, each plate 2 could have a series of axially spaced ribs which extend across the plate and, like the pads, project a discreet distance into the boundary layer 4 of the working fluid. The pads or ribs cause both the formation of a controlled vortex system, typically as depicted in Figure 6, and through heat transfer at the step-up positions additional displacement of thermal energy along the regenerator. They

also form a series of local heat sink/heat source sites on the regenerator surface which permit the thermal phase lag between the surface of each regenerator and its bulk to be increased (as will be disclosed in the context of Figure 6,) preferably to a value greater than 45° .

Instead of the pads 3, there maybe provided as mentioned earlier a series of strips or ribs (not shown) extending transverse to the directions of flow of the working fluid, ie so that in cross-section, they appear just like the pads in Figure 4.

By way of example, the regenerator plates 2 may be made of kapton, glass, quartz, silicon or ceramics and they may be less than 0.5mm thick. Meanwhile, the pads or strips may be of aluminium deposited in any appropriate manner onto the plates 2 and they may have a thickness of less than 0.5 microns, may be 0.125mm across and may be at 0.175mm centres.

The working fluid may be Nitrogen at a pressure of ten atmospheres and the local bulk movement of the Nitrogen, ie the movement indicated by the arrow 5 in each of Figures 2 and 4 may be 0.5mm or greater.

Instead of flat plates or a coil the regenerator may comprise a plurality of nested coaxial tubular plates (not shown) again with pads or strips on the surfaces.

By way of further example we will consider figures 5, 6, 7 and 8.

Figure 5 depicts a pipe closed at the left hand end and in which acoustic behaviour is stimulated by an oscillator at the right hand end.

If we consider the adiabatic behaviour of a parcel of gas situated at the position x within the pipe then when stimulated at the fundamental resonance frequency of the pipe and with the gas parcel moving through a distance S it can be shown that a temperature profile exists within the gas and a local temperature gradient of

$$-1.5708 \left(-1 \right) \frac{T_x}{L} \cot \frac{x}{2L} \text{ exists Eqn. A14}$$

ie if all heat flows are ignored gas molecules at the closed end of the pipe are hotter than those at the open end. If now heat is extracted and later re-admitted to the gas then a beneficial heat pumping condition may result although of course the actual behaviour of the individual gas parcels will no longer be exactly adiabatic. This is the concept which underlies thermo-acoustic and pulse tube cooling.

The present invention is concerned with the efficiency of this heat pumping system through the rational design of an appropriate heat exchange and temporary heat storage system.

In most of the gas flow to heat exchanger surface problems which have in the past been evaluated improved heat transfer has been effected through the generation of random turbulence within the gas much of which has been ineffectual

in the heat transfer sense and it also has the particular demerit of destroying through gas molecule collisions much of the kinetic head of the gas flow. In the oscillatory flow state conditions of thermo-acoustic and pulse tube cooling this random condition, if present would greatly reduce system performance. However in the present case no wanton destruction of kinetic head occurs and as a result a strong correlation exists in the figure 6 case between the intensity of the depicted primary vortex state and heat transfer to and from the plate.

In this context it can be shown that in extremely narrow slots the distribution of the velocity across the slot widths under oscillating motion conditions is approximately parabolic and when this is the case the energy losses are such that no useful acoustic resonance state exists. If the slot is wider then the laminar flow velocity distribution becomes progressively more rectangular in outline and of course an acoustic resonance state can then be generated and sustained. It can be shown that the onset of this beneficial acoustic state corresponds approximately to a slot width H given by the expression.

$$KH = 2.5\pi \quad (1)$$

$$\text{where } K^2 = w/2 \quad \nu \quad (2)$$

w = oscillation frequency in radians/second

and ν = th kinematic viscosity to the gas.

As an example this value of H for dry air at normal ambient pressure and temperature is approximately 1mm when the resonance state is 400 cycles/second.

As already stated it can also be shown that a system of discreet steps, as depicted in figs 6 and 7, can generate a controlled local vortex state which enhances heat transfer to and from the plate. In this context it can be shown that the energy loss per notch, or the intensity of the vortex when

$2.5 \leq KH \leq 5$ is approximately given by the expression

$$2.60 \frac{h^3}{U^2} \quad (3)$$

$$H^2 \geq 2g$$

$$\text{where } h = h \left(1 + \frac{\text{notch width}}{10.4 \text{ Rib width}} \right) \quad (4)$$

U is the peak value of the bulk fluid velocity.

H is the distance between the regenerator plates.

g is the acceleration due to gravity.

h is the rib height.

For those skilled in the art it will be apparent that with such a design effective heat exchange can be arranged to occur without destruction of the acoustic Q of this system.

Again to those skilled in the art it is apparent that the major heat exchange state occurs at the step-up location and hence the effect of gas motion can be accentuated by

constructing the ribs as heat conducting element. In fact, if we assume the ribs have heat conducting properties but have only a very small thermal storage capacity then it can be shown that, as an example, for the geometry of figure 7 the instantaneous temperature profile (in the absence of heat leakage effects) is as shown in figure 8 and the effective phase angle between heat input and storage is 50.6° . In a system which is normally considered to be loss dominated this represents a significant discovery.

As will be appreciated by those skilled in the art, for best acoustic efficiency the regenerator plates may have tapered ends and/or may have different lengths so as to reduce acoustic reflections (which may lead to the generation of harmonics). In addition, of course, the cross-sectional area available for gas flow may not be constant throughout the length of the pipe and in any case in the region of the regenerator, the transverse dimensions of the pipe may be made greater than elsewhere to allow for the presence of regenerator structures.

CLAIMS

1. A cooling device of the kind comprising a regenerator and a working fluid which is in contact with the regenerator and which is also subjected to movement, compression and expansion so as to produce a cooling effect, said regenerator defining at least one surface alongside which said working fluid passes and which has a plurality of portions projecting out into the working fluid so as to create local discontinuities in the laminar boundary layer flow of the working fluid and hence improve heat transfer both between the regenerator and working fluid, and between the working fluid and the regenerator, and further so as to create localised heat source/heat sink sites for increasing the thermal phase lag between said surface and the bulk of the member defining that surface.
2. A cooling device according to claim 1, wherein said cooling device comprises an elongate enclosure.
3. A cooling device according to claim 1 or claim 2, wherein said regenerator comprises a plurality of thin flat plates extending approximately parallel to one another and the axis of the elongate enclosure.
4. A cooling device according to claim 1 or claim 2 wherein said regenerator comprises a coil of thin material extending approximately parallel to the axis of the elongate enclosure.

5. A cooling device according to claim 1 or claim 2 wherein said regenerator comprises a plurality of nested coaxial tubular plates.

6. A cooling device according to anyone of the preceding claims wherein said portions projecting into the working fluid comprises a plurality of pads.

7. A cooling device according to anyone of the proceeding claims, wherein the cooling fluid is a gas or vapour.

8. A refrigerator including a cooling device as claimed in any one of the preceding claims.

9. A refrigerator including a chamber for being cooled and storing articles to be cooled;

an insulating layer surrounding said chamber;

a cooling device of the kind comprising a regenerator and a working fluid which is in contact with the regenerator and which is also subjected to movement, compression and expansion so as to produce a cooling effect, said regenerator defining at least one surface alongside which said working fluid passes and which has a plurality of portions projecting out into the working fluid so as to create local discontinuities in the laminar boundary layer flow of the working fluid and hence improve heat transfer both between the regenerator and working fluid, and between the working fluid and the regenerator, and further so as to create localised heat source/heat sink sites for increasing

the thermal phase lag between said surface and the bulk of the member defining that surface,

said cooling device being located in an enclosure which is insulated in the heat source areas and in which the heat sink areas are substantially inside the chamber for effecting said cooling;

and a compressor for causing fluid to pass over the regenerator.

10. A refrigerator according to claim 9 wherein said compressor is a reciprocating compressor.